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BALL BEARINGS

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ABSTRACT

SPINNING FRICTION IN THRUST BALL BEARINGS

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The spinning action of a ball with respect to the race accounts for the major part of overall friction in thrust ball bearings. New angular contact bearing applications in swivel nozzle jet engines indicated a need for research in spin friction for different bearing materials.

These tests showed the superiority of PWA-771-B as a race material against a 440 C ball bearing from the criteria of minimum friction. Evidence of accelerated fretting corrosion due to the presence of oxygen points out the importance of excluding oxygen in the bearing. The spin friction coefficient was not constant as assumed in the ideal model of spinning. Further research is necessary to propose an improved model for spinning friction at high Hertz stresses.

Thesis Supervisor: Walter D. Syniuta
Title: Professor of Mechanical Engineering

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INTRODUCTION

Previous research has shown that the spinning action of a ball with respect to the race accounts for the major part of the overall friction in thrust ball bearings. This spinning action of the ball is about an axis perpendicular to the area of contact between the ball and the race. In high speed thrust ball bearings, the spinning is due to the interaction of centrifugal forces and thrust forces. This causes the contact between the outer race and the ball, and the inner race and the ball to differ across the diameter of the ball. This difference in contact across the diameter results in the spinning of the ball about one of the contact areas. In angular contact thrust ball bearings, the spinning action is due to the geometrical arrangement of the bearing.

The friction that results is similar to sliding friction, the difference being that the ball material is revolving and the relative velocity in the contact zone is proportional to the distance from the center of contact. Also, the stress above the contact area is not uniform.

New angular contact bearing applications in swivel nozzle jet engines indicated a need for research in spin friction for different bearing materials. A description of the bearing is as follows; cross section, 0.75" x 0.75"; ball diameter, 0.375"; shoulder height, 33% of ball diameter; proposed materials, 440C modified, both ball and race. The operating environment is: Maximum race temperature, 500°F, maximum race-to-race temperature differential, 50°F, atmosphere, gas turbine exhaust gases. The maximum bearing Hertz stresses are 400,000 psi and the bearing is required to oscillate a maximum of 180°. It is anticipated that

a high material hardness will be required to resist brinelling under high static bearing loads.

APPARATUS AND EXPERIMENTAL PROCEDURE

An overall view of the apparatus is shown in Figure 1, and a close-up of bearing arrangement is shown in Figure 2. The equipment consists of four sections: the drive-motor (A), the connecting linkage and strain ring (B), the test section (C), and the dead weight load assembly (D). Auxiliary equipment consisted of a Sanborn Carrier Preamplifier Model 150-1100 AS, and a desiccator for bearing storage.

The drive motor is a constant speed, single phase, AC motor with an output speed of 60 rpm. The linkage connections were adjustable so that the clearances could be minimized, and the stroke was adjustable. A full bridge circuit was used on the strain ring. The test section is comprised of three plates (A,B, and C) and seven ball bearings. The upper plate (A) and the lower plate (C) are stationary, while the middle plate (B) is driven in an oscillatory motion through a rigid connection with the strain ring (D). Three 3/8" ball bearings separate the upper and middle plates, and roll on hardened pads made from 52100 stock. The mounted pads were ground to a flat surface which was polished to a mirror finish with 4/0 emery paper. The middle and lower plates were separated by one similar pad arrangement, and two ball bearings (E and F) rolling in V grooves (included angle - 120°). The grooves were made out of the desired race materials, and only one set was used so that the apparatus was self aligning. To take up small misalignments, and ensure approximate uniform distribution of load in the test section, a single ball bearing was placed between the upper plate and the housing. This ball was seated in an appropriate sized washer to prevent its motion. The dead

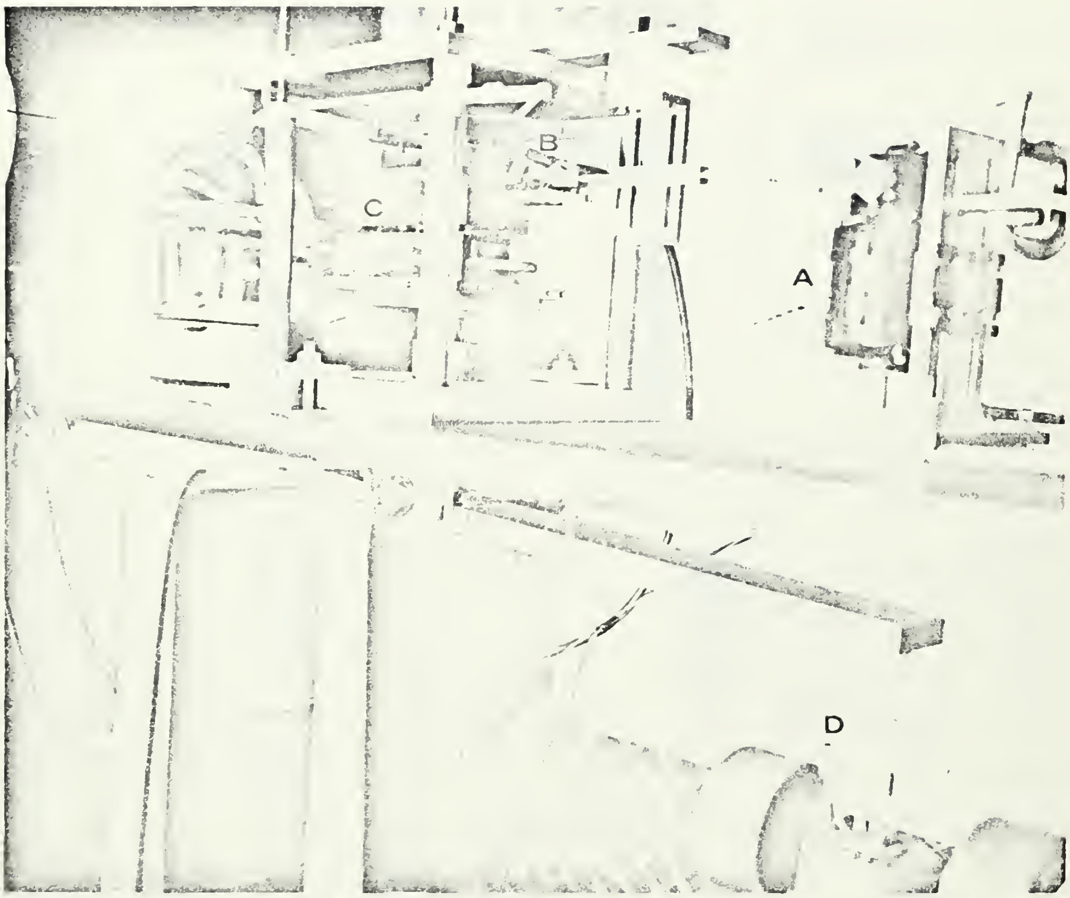

Figure 1. Spin friction apparatus

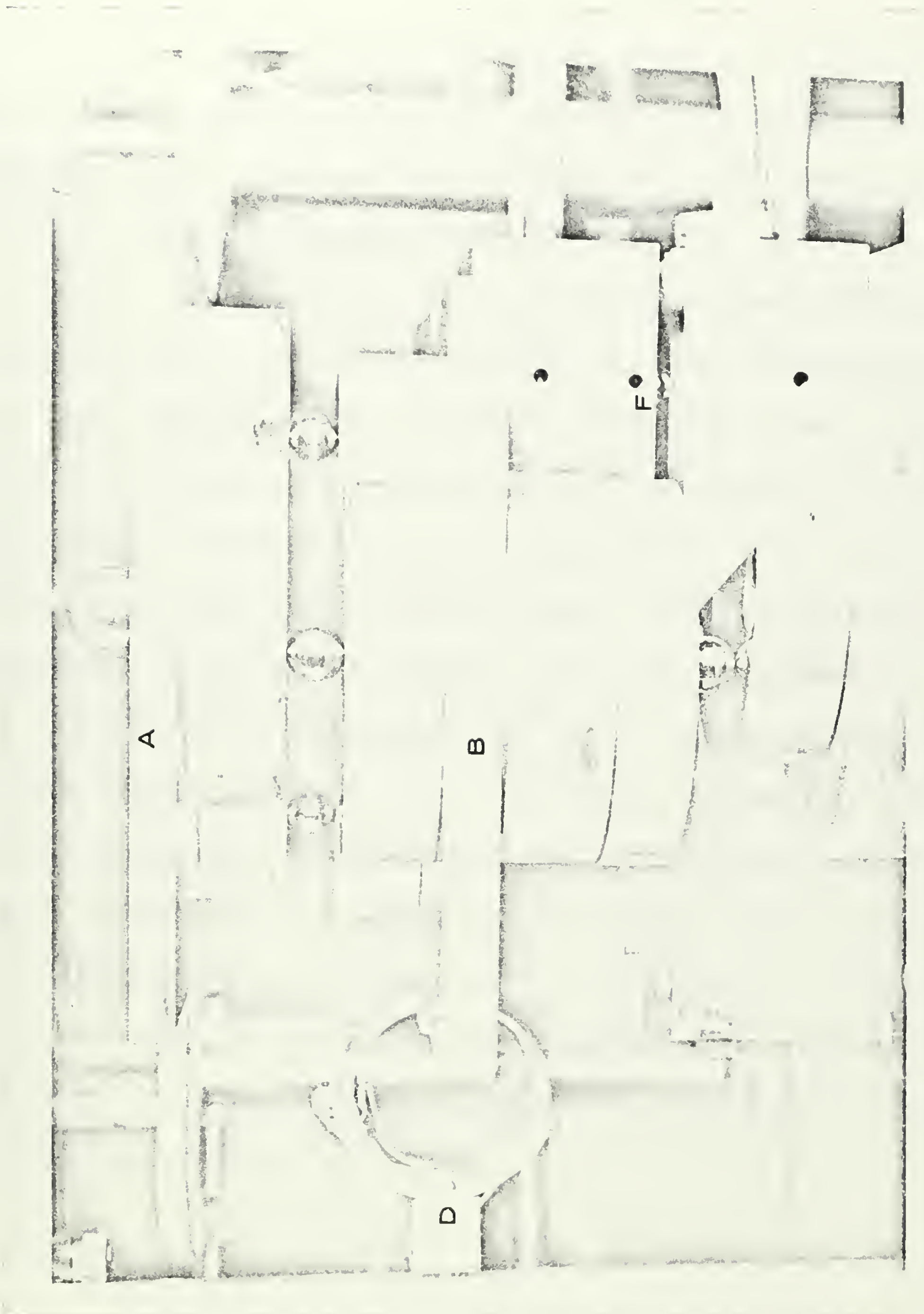

Figure 2. Close-up of bearing arrangement

weight load assembly consisted of: a weight pan, 10 to 1 lever arm, and a column on which the test section was placed. This column moved vertically in a linear bearing, and was separated from the lever arm by a ball bearing. The wooden table which supported the housing and leveling blocks helped to damp vibration.

The balls used in the testing were 3/8 inch diameter 440C (martensitic stainless) of ball tolerance grade 25. The race materials tested were: 440C, 5630 C, PWA 771-B, E52100, and M2 (tool steel). After the balls were degreased with trichloroethylene and washed with laboratory glass cleaner, they were stored in the desiccator. The rails were polished with 4/0 emery paper before testing. To minimize the effects of deformation, all tests were started in the light load condition. Weights were then added every 30 seconds until the full load condition was reached (Hertz stress of 400,000 psi). The Sanborn recorder was set to a sensitivity of 4 gm/mm., and the strain ring was calibrated with a 100 gm weight. The stroke for all tests was 9/32 inch. Since the displacement of the middle plate was sinusoidal, the average absolute velocity was 2.81 ft/min. and the maximum inertial force was 68.1 grams at the end of each stroke. The testing was done in an air conditioned, vibration isolated room at a temperature of 68°F.

RESULTS AND DISCUSSION

The measured moment at each ball-race contact in the V-groove is equal to the sum of rolling moment and spin moment. It was originally intended to include measurements of rolling moment by testing the balls on flat pads. These tests were discarded for two reasons. The first was that a transverse vibration of the middle plate across the strain ring interferred with results. The second reason was that a spiked peak near the end of each stroke distorted the results. The magnitude of the rolling moment is much smaller than the spin moment and the neglect of this measurement did not introduce appreciable error.

The Sanborn Pre-amplifier recorded the force applied to the middle plate. Due to vibration in the apparatus, it was necessary to use damping on the recorder pen with a time constant of 0.1 second. The calculated inertial force of the middle plate for the stroke in these tests was 68.1 grams. The forces recorded ranged from 62 to 65 grams.

A series of tests were run to test the influence of the manner of applying load. A run was made in which the weights were applied until the maximum load was reached. The weights were then removed with the apparatus still running. It was found that the forces measured while decreasing the load were greater. Another run was made from maximum load to minimum load with new balls in a new position on the V-grooves. These force readings were also greater than the readings taken from minimum load to maximum load. Figure 3 shows a ball bearing in a V-groove after one test with the load increased during the run. The groove in the rail (A) is typical of all tests. The wear on the ball

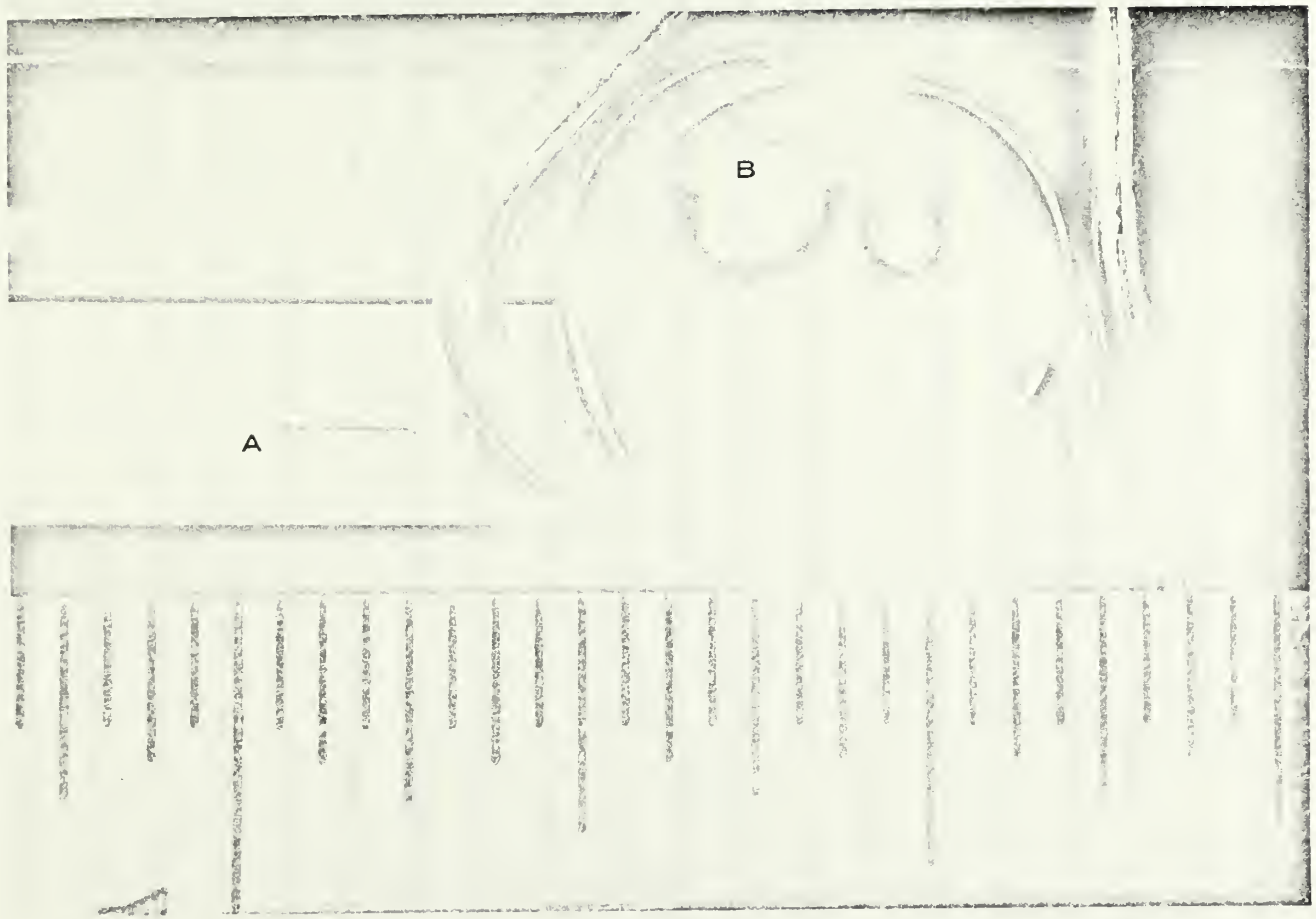

Figure 3. Ball and rail close-up (approximately 7 X) showing fretted areas after testing.

(B) is not extensive, but is readily apparent due to the presence of iron oxide.

Accurate results could not be obtained below 200,000 psi Hertz mean stress due to high inertial force magnitudes. Thus essentially all tests were done in the plastic region. Although the yield stress for each specimen is not known, the yield stress for the 440 C materials (440 C and 5630 C) is certainly no greater than 275,000 psi. The maximum Hertz stress is approximately 50% greater than the Hertz mean stress. This implies that plastic deformation was present in all the testing. The observations regarding the manner of loading are then readily explained by the hysteresis that occurs when changing loads in the plastic region. Prof. H.H. Uhlig's model of wear due to fretting corrosion includes both wear due to the rubbing of loaded surfaces, and the accelerating affect caused by the presence of oxygen. Iron oxide was observed on both the ball and rails, but none on the pads where only rolling occurred (no slip condition). This emphasizes the need for some other means of excluding oxygen in the actual bearing to extend its useful life. The use of a low viscosity oil is effective since it penetrates into the contact between the ball and race more readily than a high viscosity oil, and retards the diffusion of oxygen to the bare metal surface produced by wear. It also raises the question of the validity of results, especially at the higher stress levels. These measurements were taken last, and it might be expected to have oxide to oxide contact rather than metal to metal contact. Additional testing in a nitrogen atmosphere to ensure metal to metal contact would be

necessary to investigate this point further.

The waveform of force versus time should have been a square wave. This would represent Coulomb friction independent of velocity and a relatively small inertial force. The actual wave form contained a spike of the same order of magnitude as the friction forces. It occurred near the end of each stroke and disappeared with the reversal of direction. In observing the results at the lightest load condition, the sinusoidal wave form of the inertial force was readily observed. After completing a run, and returning to this load condition, the spiked wave form was apparent. Thus, the spike is attributed to the formation of the groove. Since the groove is shallower at the ends, it therefore requires more force to elastically deform this portion of the groove. At higher stress levels the spike was observed to increase with time implying plastic deformation also.

Figure 4 shows the graph of spin moment versus normal load. The spin moment should vary as the normal load to the 4/3 power based on a constant spin friction coefficient and a normal Hertz stress distribution. In general, the slopes are greater than 4/3 except for 440 C. Some of the slopes appear to steepen in the higher load range, perhaps due to formation of oxide layers. The tendency to form increased amounts of oxide is stronger at these high loads due to the increased wear. A layer of oxide would then accumulate during each test. The extent of metal contact would decrease, and the slope would change due to the new ball-race contact conditions. Figure 5, the graph of spin friction coefficient is not constant. This again may indicate a change in mechan-

ism, such as oxide to oxide rubbing. The sources of error in the measurement include over damping and the neglect of rolling friction. These errors are opposite in sign and the combined error is on the order of 5%. Another problem is that of repeatability of results. Certainly the vast difference between 440 C and 5630 C, essentially the same material at the same hardness, raises some doubts. This class of stainless is known for its widely varying yield point at these hardnesses, and this may have been the difference between the two tests. The one outstanding feature of these results is the clear superiority of PWA 771-B. This material's composition is very similar to that of the 440 C stainless, except for the addition of 10% tungsten, 10% Cobalt, and a higher carbon content. It may be desirable to conduct further tests to see if these results can be correlated with changing the proportions of these elements.

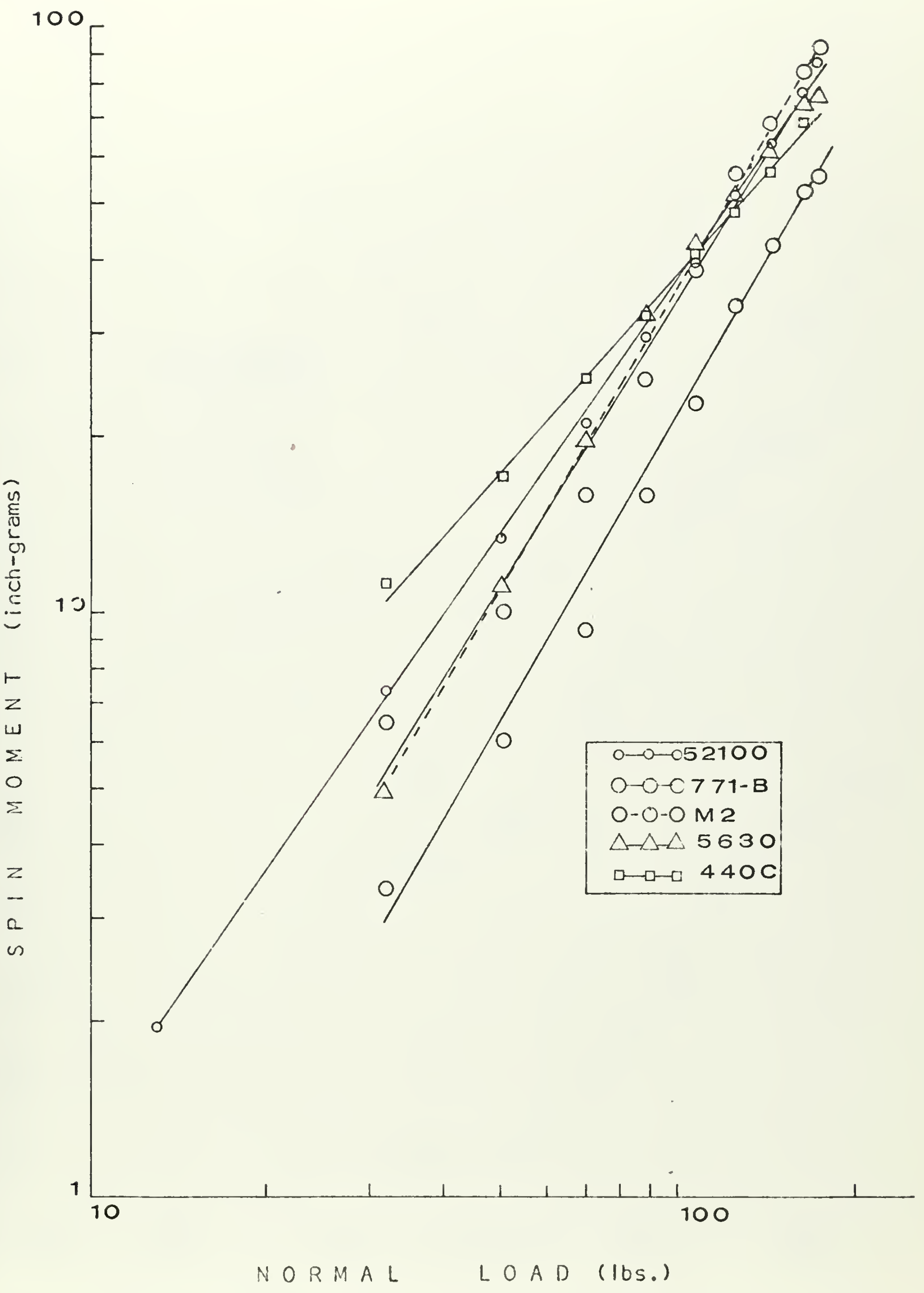

Figure 4. Graph of log spin moment versus log normal load at contact.

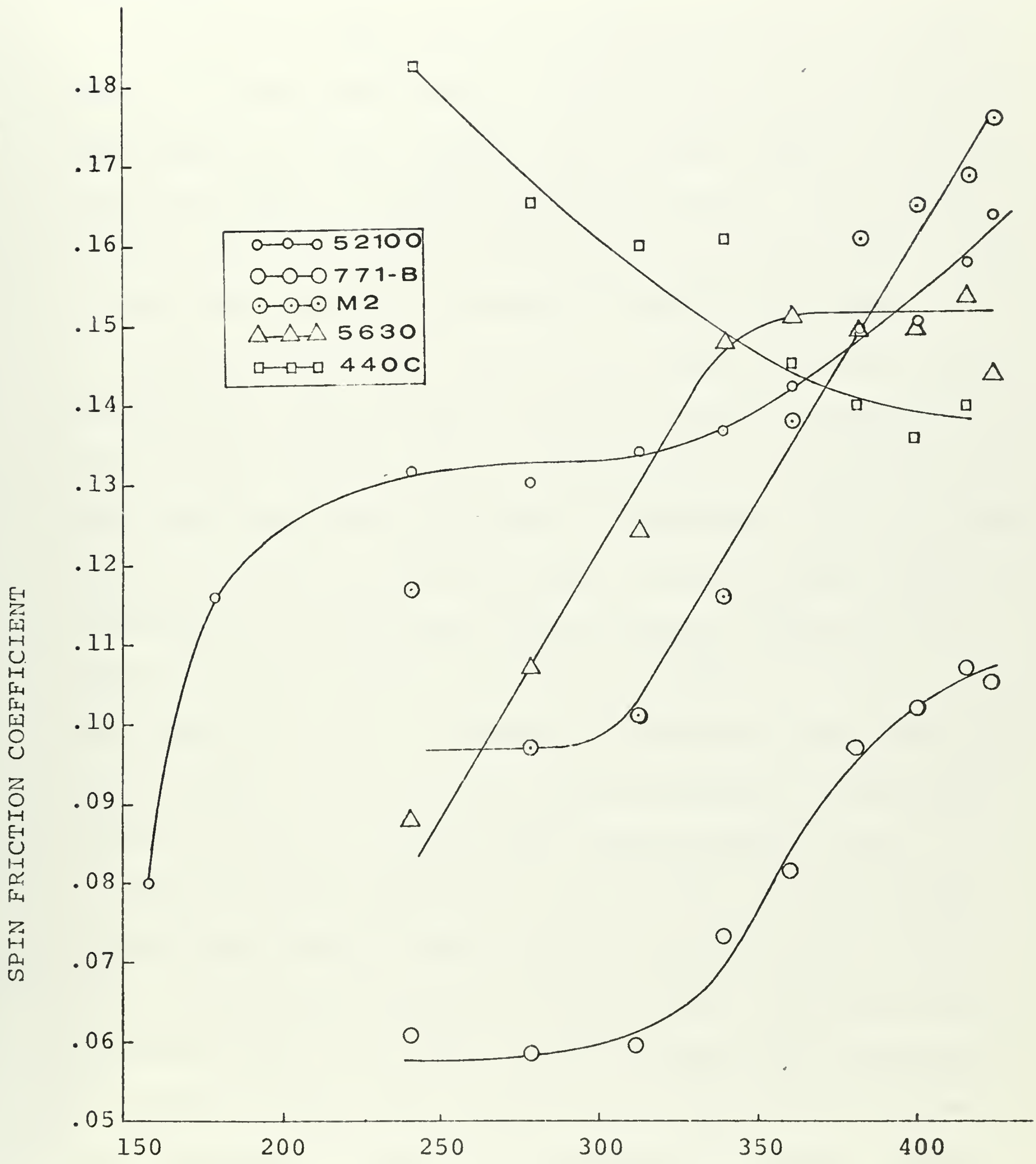

Figure 5. Graph of spin friction coefficient versus Hertz mean stress.

CONCLUSIONS

The object of these tests was to survey materials to aid in the design of a nozzle thrust bearing. The following race materials were tested against 440 C ball bearings: 440 C, 5630 C, PWA 771-B, E52100, and M2 (tool steel). PWA 771-B was clearly superior. This material is similar in composition to class 440 C martensitic stainless, but has a higher carbon content, and additions of tungsten and cobalt.

Fretting corrosion did occur during the tests due to the rubbing of highly loaded metal surfaces, and wear was probably accelerated by the presence of oxygen. There was some evidence that the mechanism may have changed above a Hertz mean stress of 370,000 psi. This could have been due to the build-up of oxide layers during the testing. The use of some means to exclude oxygen from the bearing is important to extend the life of the bearing. One method of doing this is to use a low viscosity oil. It will penetrate into the contact areas more readily than high viscosity oils and is effective because it reduces the diffusion of oxygen to the bare metal surface produced by wear.

The spin friction coefficient is not constant. This points out the need for an improved model to explain the metal to metal contact results in this type of testing. One explanation, due to the increasing slope in four of the tests, is that the results were distorted by the build-up of oxide layers. On the other hand, deviation from the basic assumptions of a Hertz stress distribution and elastic deformation may explain these results.

RECOMMENDATIONS

This was the first series of tests made with this apparatus. Ghiya had done previous tests with a similar apparatus, but had used much larger diameter balls (7/8 and 1.25 inches). His tests were done with various oils to test their influence. NASA has constructed an apparatus based on a different testing scheme. The ball is held in a chuck, and loaded against a race material. The test is conducted by spinning the ball and measuring the resulting torque. The spin friction coefficient is calculated by using the ideal spin moment equation based on a 4/3 slope. Presently, only preliminary tests have been conducted with lubricant using 52100 steel and a race conformity of 51 percent.

In this series of tests the stroke was made relatively short in order to reduce the magnitude of inertial forces. Unfortunately, this gave relatively poor results due to the unexpected spike at the end of each stroke. In future tests, the stroke should be lengthened and the drive motor speed reduced. These changes will eliminate and possible influence of over or under damping by lengthening the time of measurement. This, also, should improve the results of the rolling friction tests, and enable further refinement in the calculations.

Further investigations should include an effort to determine the repeatability of results. The addition of a counter to record the number of cycles and varying the number of cycles at each load setting may increase the understanding of the factors involved. Also, the inclusion of a tensile specimen with each set of race material would give additional useful data. Another facet to this same question includes the

relation of sliding friction coefficient to the spin friction coefficient. A simple set of tests to obtain these results could be valuable in analyzing spin friction data. Further parameters that need investigation include temperature effects, ball diameter, alloy content, hardness, and the effect of fretting corrosion due to the presence of oxygen.

A different approach to these problems would be the construction of an apparatus similar to NASA's. This type of test looks directly at the problem as hypothesized, and is necessary before further investigation of elastic compliance (a region of no slip that occurs near the leading edge of the contact area under certain conditions of rolling and spinning). Also, the small size and simplicity in such a test should enable a vast number of materials and conformities to be tested with ease, and reduce the problems involved with high temperature or nitrogen atmosphere tests. The assumption that the load is uniformly divided between three ball bearings would no longer be necessary, and the time involved in rigging the apparatus and checking the alignment would be saved.

APPENDIX

A. SAMPLE CALCULATIONS

B. SPECIMEN PROPERTIES

SAMPLE CALCULATIONS

Symbols:

P	The force required to oscillate the test section
M	Measured moment at ball and V-groove contact
M_r	Friction moment under rolling
M_s	Spin friction moment
a	Hertz radius of contact between ball and flat surface
N	One third the load on the apparatus
N'	Normal load at a ball and V-groove contact
μ_s	Spin friction coefficient
θ	$(180° - 120°)/2 = 30°$ Angle of contact
$120°$	Included angle of V-groove
$D =$	3/8 inch Ball diameter
$E =$	30×10^6 psi Young's Modulus of Elasticity

Calculation:

Resolving spin and rolling moments at the contact of the ball and V-groove,

$$M = M_s \sin \theta + M_r \cos \theta$$

where $M = P\, D \cos \theta/4$

and $N = 2\, N' \cos \theta$

In the case of a ball in contact with a flat surface,

$M_s = 3\pi N' a\, \mu_s/16$. Since M_r was not measured and is relatively small, this term was assumed to be zero. If this moment was used, the force P would have to be correct for the 4 balls rolling in each test.

Example:

Material: PWA 771-B

Diameter: 3/8 inch

Normal Load on apparatus = 361.7lb

$N = 361.7/3 = 120.6$

$N = 2\ N' \cos\theta$

$120.6 = 2\ N' \cos 30°$, $N' = 69.6$ lb, $N'^{(1/3)} = 4.11$

The force required to move the middle plate as read the Sanborn recorder (P) was 115 gm.

$M = DP \cos\theta/4 = 3P \cos 30°/8 \times 4 = (0.0812)P$

$M = 115 \times 0.0812 = 9.34$ inch gm

Since $M = M_s \sin\theta = 3\pi N' a\ \mu_s/16$, and from Hertz's formula,

$$a = 0.881\ \left(\frac{N'D}{E}\right)^{1/3}$$

therefore $\mu_s = 1.828\ M/N^{4/3}$

$$\mu_s = 0.0596$$

The Hertz mean compressive stress equals $0.410\ \left(\frac{N'E^2}{D^2}\right)^{1/3}$

$$\text{STRESS} = 76{,}100\ N'^{(1/3)}$$

$$\text{STRESS} = 313{,}000 \text{ psi}$$

SPECIMEN PROPERTIES

	440 C	5630 C	PWA 771-B	E52100	M2
Carbon	.95-1.20	.95-1.20	1.15-1.40	.95-1.10	.85
Chromium	16-18	17	17-18	1.3-1.6	4
Molybdenum	max .75	.50	max .50		5.00
Tungsten			10-11		6.25
Vanadium			.5-1.		2
Manganese	max 1.00	max 1.00	max .75	.25-.45	
Silicon	max 1.00	max 1.00	max .75	.20-.35	
Phosphorus	max .040	max .040	max .04	max .025	
Sulphur	max .030	max .030	max .04	max .025	
Cobalt			9-10.		
Boron			max .010		
Nickel	max .50	max .50	max 1.00		
Copper			max .50		
Titanium			max .10		
Hydrogen			max .010		
Oxygen			max .100		
Nitrogen			max .075		
Rockwell C	57	57	56	58	60

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